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THE SONIC VELOCITY SLIPPAGE CONCEPT FOR RATING THE VOLUMETRIC EFFICIENCY OF ROTARY COMPRESSORS

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INTRODUCTION

This paper offers a procedure of evaluating the performance of rotary compressors. A rotary compressor is a machine used to elevate the pressure of gas. The operation usually involves ports, rather than valves per se', to control the gas flow. The motivation is the result of a rotating core within a fixed and confining cylinder. These rotors have various configurations as shown in the appendix figures.

DISPLACEMENT

Ref 8

The basic displacement of the machine is the product of the rotary diameter (d), its cross-sectional area factor (X), the rotor length (L) and the rotor tip speed (U). The equation reads:

$$QD = d\ L\ U\ X \quad (1$$

$$A1 = 0.004\ L + 0.3d/nc \quad (2$$

$$A2 = 0.06\ L + (0.6d/nc) \quad (3$$

$$A3 = 0.06\ L + (0.125d) \quad (4$$

$$QS = 35\ A * SQR\ (k * T2/m) * R\uparrow\sigma \quad (5$$

APERTURE & SLIPPAGE

Equations 2, 3 & 4 define the aperture areas of the escape passages that exist between the discharge and the suction chambers of the compressor casing. The numerical constants relate to the blade and end clearances. Such data is considered proprietary information by the manufacturers. Eq 2 relates to the sliding-vane type compressor with nominal end clearances and minimal vane-cylinder contact. The maximum contact clearance can extend the constant "0.004" to "0.040". The small letters "nc" refer to the number of individual cells contained in the periphery.

Eq 3 is a proposed measure of the aperture area related to the HELICAL SCREW and the SPIRAL-AXIAL type compressors. Eq 4 relates to the STRAIGHT LOBE type compressor. A clearance of 0.020 inch was given for the popular sized 11 inch diameter rotor and is used in Eq 3 and 4. The letters QS refer to the SLIPPAGE flow in icfm. $R\uparrow\sigma$ is the thermal change, exponential-factor. No SLIPPAGE equations are proposed for the LIQUID-LINER COMPRESSOR, SEE CLOSURE.

VOLUMETRIC EFFICIENCY

If a perfect seal existed at all points of contact, the compressor has 100% volumetric efficiency, zero slippage and the capacity is equal to the displacement. If the compressor is operated against a closed discharge, it has zero-flow and 100% SLIP.

The volumetric efficiency is:

$$Evr = 100 - (QS/QD)*100 \quad (6$$

CHARGING & EXHAUST EFFORT

Ref 2& 4

Energy is spent for every change in the flow pattern of the gas machine. The charging of each rotating cell requires about two velocity-heads (VELADS) in terms of the rotor tip speed. Ref 1. It requires one VELAD to exhaust the cell. The dynamic losses are about 0.5% for rotor tip speeds less than 0.05 MACH. These losses amount to 5% of the system pressure for tip velocities $\leq$0.1 MACH. The dynamic losses are negligible for all rotary compressors, except for the HELICAL-SCREW type units where the rotor tip speed exceeds 0.2 MACH. The suction and discharge losses are determined from the following procedure:

$$\theta 1 = 2.5\ m\ U\uparrow 2/T1\ (10)\uparrow 5 \quad (7$$

$$\theta 2 = \theta 1/R\uparrow\sigma \quad (8$$

$$B = (1 + \theta 2)/(1 - \theta 1) \quad (9$$

$$Edy = (R\uparrow\sigma - 1)/(BR\uparrow\sigma - 1) \quad (10$$

$$W = Q/V1 \quad (11$$

HEAD & POWER

The gas head and the power required to raise the system pressure is given in the following equations:

$$Had = (R\uparrow\sigma - 1)\ 772.5*(Z1 + Z2)*T1/m\sigma \quad (12$$

$$Hise = 778\Delta Enthalpy\ (P2 - P1) \quad (13$$

$$Ad\ hp = H*\ W/33{,}000 \quad (14$$

$$Ise\ hp = W*\Delta H/42.5 \quad (15$$

$$Gas\ hp = Adhp/Evr*\ Et*\ Edy \quad (16$$

$$Bhp = Gas\ hp + Gas\ hp\uparrow 0.5 \quad (17$$

The adiabatic head is developed in Eq 13. When enthalpy tables or Mollier Charts are used, this text refers to such changes as isentropic. The delta H in Eq 14 & 16 represents the isentropic change between suction, to discharge conditions. This satisfies the reversible requirement of the Second Law of Thermodynamics. The SLIPPAGE gas must be recompressed. Hence, Evr has a place in the denominator of Eq 17. An allowance must also be made for the "warm-up" suction temperature (Et) caused by the SLIP. The dynamic losses plus the two above losses relate to the Gas hp in Eq 17. An allowance equal to the SQR (square root) of the Gas hp is usually a generous figure for the mechanical losses of the unit including necessary gearing.

SIZING & RATING

A typical sliding-vane type compressor's application to the following conditions is developed below from the procedure just described.

TABLE I

Condition:		Temp	Pressure
Suction; T1 & P1	(60 F)	520 R	14.7
Ad Disch; T2# & P2	(252 F)	712 R#	44.1

Hardware; d = 10.6 in., L = 15.5 in. or to suit, X = 0.0617, N = 1175 rpm, U = 54.5 fps, R = 3.0, k = 1.40, σ = 0.286, R^{σ} = 1.37# m = 29, W = 35 lb/min, τ = 1310 fps, H = 35,800 ft, Ad hp = 38.

	MIN	MAX
Aperture area	0.338	0.892
QS, SLIP, icfm	96	254
Q, Capacity Required	459	459
QD, Disp Required	555	713
L, Length Required, in.	15.5	20.0
SLIP, %	17.3	35.6
Evr, Vol Eff, %	82.7	64.4
Warm-up Suction, °F	91	127
Mixture Discharge °F	295	345
Et, Warm-up, Eff	94.5	88.5
Edy, Dynamic Eff	99.2	99.2
Gas hp	49	67
Bhp	56	75
Overall Efficiency %	68	51

The above table gives the contrasting performance of a popular sized sliding-vane compressor, at minimal and maximum aperture conditions. To compensate for the larger aperture, the rotor length must be extended 30% and the power increased 33%. The efficiencies are not impressive. The "mixture discharge" temperature is the most interesting feature. It makes an excellent performance indice.

HELICAL SCREW COMPRESSORS

The helical screw compressor is the only rotary unit that operates at tip-speeds in excess of 0.1 Mach. Table 2 shows five distinctly different categories of rotary compressors. The spiral axial operates at 0.1 Mach and the other three operated at or about 0.05 Mach. The latter group can be categorically assigned a composite efficiency of 90 percent, except for the sliding-vane and the liquid-liner unit when operating above 3 R_c.

The helical screw compressor is the most versatile form of rotary compressors. A profile of the 4-lobe male and the matching 6-lobe female configuration is shown in Fig.2 A typical performance chart is shown in Fig.3. of reference (4). The optimized performance of the helical compressor can be projected on a Baljé chart. The optimum performance of a centrifugal compressor lies along a D_s N_s slope of 150 and a pressure coefficient of 0.55, with the best operating point (BOP) at D_s = 1.5 and N_s = 100 as shown on Fig.4. The optimum performance of a helical unit has a D_s N_s slope of 26 with the peak performance island at D_s = 2.0 and N_s = 13. The D_s N_s slope of 26 is identified as having a pressure coefficient of 4 [Reference (5)]. This means that the helical machine is capable of attaining the same head as a centrifugal compressor at (0.55/4) 14 percent of the centrifugal tip-speed. A reciprocal statement would read that the helical unit could develop 7.3 times the head realized from a centrifugal compressor operating at the same tip speed. This low specific speed characteristic is very desirable for compressing small volumes of gas to high pressures.

The most successful helical screw applications are delivering air for general construction, road building purposes and used mainly for operating pneumatic tools. The size of these portable units run from 600 to 1200 icfm of 125 psig air and require 250 to 500 hp. The key to their success is the moderate tip speed of 0.11 Mach and lower. The second factor is the technique of internal oil flooding. The intake is sprayed with oil at a ratio of one part per thousand, or 7.5 gallons per thousand cubic feet. This flooding absorbs about two-thirds of the heat of compression, reducing a normal 350 F discharge temperature to about 100 F above ambient or 170 F. This eliminates the need for an after-cooler. The oil is recovered from the separator, cooled, and returned to the compressor suction. The sheep-skin filters are most efficient, less than 0.1 percent of the oil (one gallon per 60,000 hp-hr) is lost. The oil also functions as a lubricant and a sealant to minimize the by-pass leakage. The latter is perhaps the most important function of all in that it silences the shrill by-passing noise.

The helical unit was orginally considered to be only suitable for dry gas. Condensate and grit particles were taboo. These restrictions have been relaxed in recent years. These machines are being used to handle the waste gas from a hydrogenation process. This is a most difficult service where occasional blobs of a very viscous coagulate must pass through the machine. The compression is not without its problems, but helical units perhaps have less difficulty than those which would be experienced from a piston machine in the same service. The helical machine can be operated without a driving gear. The lobes of the rotors do not experience a locking interference, which does occur in other lobe models. Timing gears are used to drive and also avoid this interference. The male lobe is the driver. The female rotates in the opposite direction, see Fig.2. The containment wrap requires 300 degrees of rotation. The pitch of the helical screw is about 45 degrees.

BUILT-IN CLEARANCES

The transition of the Lysholm design helical screw unit to the present Svenska Rotor Maskiner (SRM) form included the "built-in" clearance feature. The space between the male lobe and the female recess, as illustrated in the cross-sectional end view of Fig.2, is the "built-in" clearance. The gas retained in this clearance space is returned to the suction chamber and should balance with the suction chamber pressure. An excess pressure is equivalent to additional by-pass. An undersizing permits a back-flow into the "pocket" and requires the additional burden of pumping it back into the suction chamber.

The Svenska Rotor Maskiner (SRM) owns the patents for this machine. They have numerous licensee throughout the world. Their cast-iron housings are limited to 250 psig for 16.5 in. and smaller rotors. Larger impellers are limited to 125 psig. The housings are not available in ductile iron or cast steel. Perhaps the most suitable and neglected application is its potential usage as a gas motor or still better as a cryogenic expander. Referring to the Baljé chart where the helical compressor had a D_s N_s characteristic slope of 26 and q_{ad} value of 4, the equivalent rotary expander key performance factor (tip speed/spouting velocity) should be 0.08. This is equivalent to an 8.5 inch rotor running 3600 rpm matching the efficiency of a 4.5 inch radial in-flow impeller turning 51,000 rpm.

SPIRAL AXIAL COMPRESSORS

The spiral axial compressor has a top speed equal to the slowest helical screw machine. A cross-sectional profile of the spiral machine is shown in Fig.5. The "pocket" volume is more than twice the capacity of the helical machine having the same d, L and N dimensions. The machine has a timing gear drive and runs dry, without the rotors making contact. It provides 20 to 30 psig air for manufacturing and food processing operations.

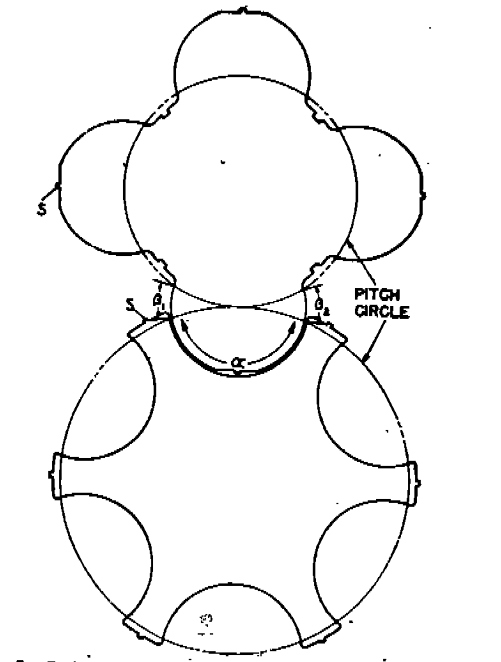

Fig.2 Rotor profiles of the 4-lobe male and the 6-lobe famale. The "built-in" clearance exists between the two rotors, courtesy of Fairbanks Morse, of the Colt Industries

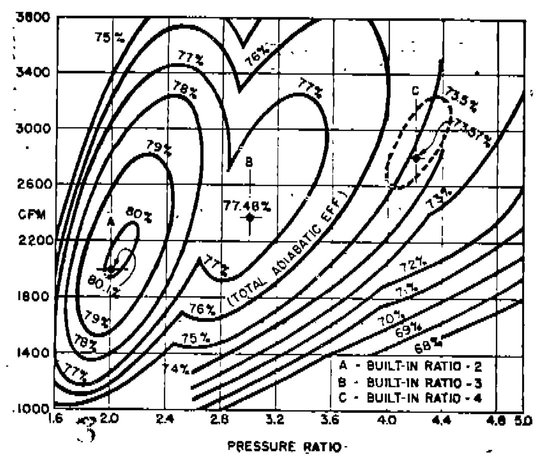

Fig.3 A typical performance chart for a helical-screw type compressor illustrating the effect of three "built in" clearances. A speed overlay pattern of horizontal sloping lines would show 3500 rpm intersecting 1400 icfm, 5500 rpm would intersect the center of the 2 R_c island, 6500 rpm through 3 R_c island and 7500 rpm through 4 R_c island. Courtesy of Fairbanks Morse, of the Colt Industries

SLIDE-VANE COMPRESSORS

The slide-vane compressor is illustrated in Fig.7. The rotor runs eccentrically within a cylinder. Radial slots in the rotor carry sliding vanes which form a series of longitudinal cells. The centrifugal spin of the rotor holds the vanes snug against the cylinder wall. The cell volume diminishes as the rotor approaches the discharge chamber, which is in the lower right hand corner of Fig.7. A single-stage slide-vane compressor can pull a 28 in. Hg vacuum or pump 50 psig. A two-stage unit can pump 250 psig air. The unit is not suited for handling saturated and super-saturated vapor.

Table 2 Rotary compressor performance data

Type	Helical screw	Spiral axial	Straight lobe	Sliding vanes	Liquid liner
Features	4x6	2x4	2x2	8 blades	16 sprockets
Max. *OD*, icfm	20,000	13,000	30,000	6000	13,000
Max. *d*, in.	25	16	28	33	48
Min. *d*, in.	4	6	10	5	12
Limiting, y Mach	0.35	0.12	0.05	0.05	0.06
Normal, y Mach	0.24	0.09	0.04	0.04	0.05
Normal, overall η	75	70	65	70	50
Normal Re	2-4	3	1.7	2-4	5
V Factor for Evr	3	3	5	3	3
X Factor for QD	0.0612	0.1333	0.270	0.0617	0.071

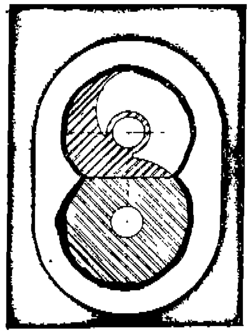

Fig.5 End view of the spiral-axial compressor showing the gas envelope

STRAIGHT LOBE COMPRESSORS

The straight lobe rotary compressor operates at the relatively low tip speed of 50 fps. This is tantamount to a 10 in. rotor driven by a 6 pole motor. An animated view of gas flow through the machine is shown in Fig.6. This machine can accommodate a considerable quantity of liquid, perhaps as much as 50 parts per thousand.

The public utilities use the straight lobe compressors as displacement meters, low-pressure pipe-line boosters, vacuum gas-well gathering systems, and as a supercharger for gas or diesel engines. This type of compressor is perhaps the most efficient form of vacuum service.

Optimum single-stage performance is realized at 20 in. of mercury and 28 in. for two-stage operation. It is used for desalinization and other evaporative processes where large volumes of vapor are removed at high vacuums.

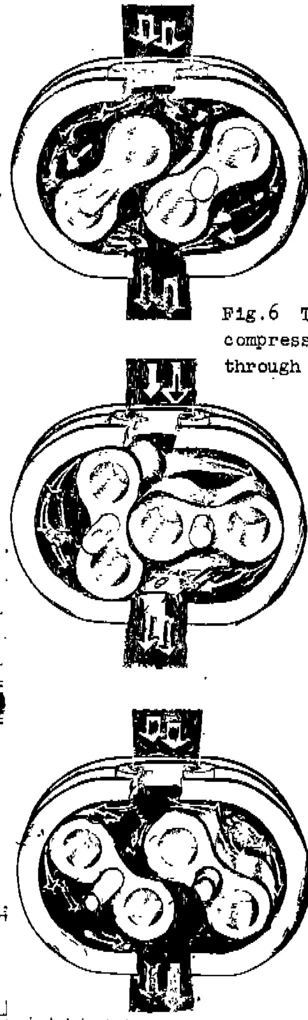

Fig.6 The three positions of a two-lobe rotary compressor give an animated view of the gas flow through the machine

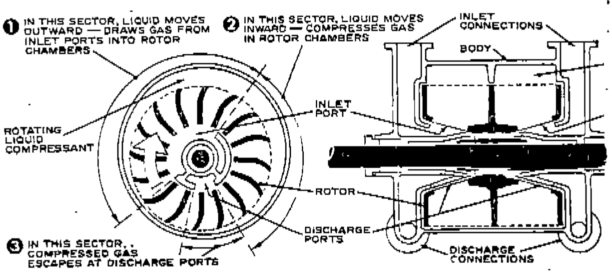

Fig.8 A sectional and end view of a liquid-liner type compressor, courtesy of the Nash Engineering Company

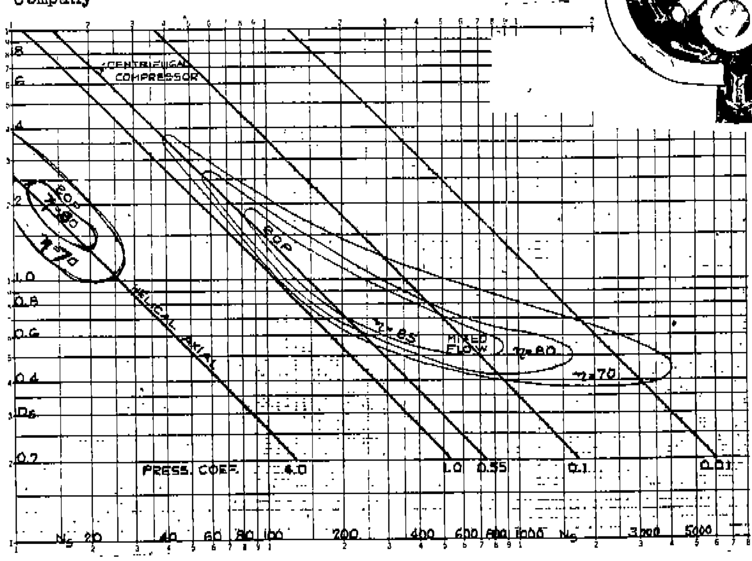

Fig.4 Baljé compressor performance chart

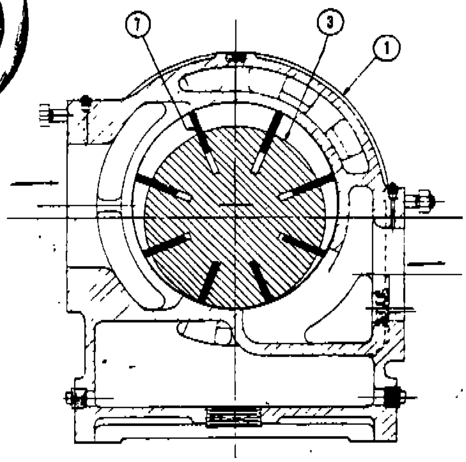

Fig.7 End view of a sliding-vane compressor. rotor turns clockwise closing the annular cells into the discharge chamber. Courtesy of Allis-Chalmers Manufacturing Company

LIQUID LINER COMPRESSOR

The operation of the liquid-line rotary compressor is illustrated in Fig.8. The sprocket blades throw the sealant (usually water) in a cylindrical form against the internal housing.

This cylindrical liner contains the gas flow from the large eccentric entrance to the confinement of the discharge port. This compressor is used for handling the more corrosive and volatile gases in chemical, textile, paper pulp and nuclear power plants. It is indispensible for handling exothermic gases like chlorine, oxygen and acetylene. The former is sealed with 97 percent concentrated sulfuric acid. Hydrogen chloride is sealed with Monsanto's AROCLOR. The liquid-liner compressor is suited for handling liquid priming and fibrous material carry-over. It is used for vacuum fibrous-plastic molding where material carry-over would interfere with lobe rotation.

The liquid-liner compressor offers an absolutely oil-free, 125 psig single-stage air compression service. It is used for hyperbaric surgery environment, for laboratory sterilization, autoclave impregnation, and to pull large evacuations as low as 25 in. Hg. Despite a miserable 50 percent efficiency, it is still ten times more efficient than a steam ejector and is being used more extensively as a replacement. It is used to remove inert gases and condensate from steam heating plants and condenser systems.

The compressor requires considerable quantity of cold coolant to maintain the discharge within 20 F of ambient or the feed temperature. The manufacturers advocate complete rejection of all mechanical energy expended for the compression. This is an exaggerated requirement and a rather costly auxiliary service, especially when the coolant is not permitted to rise above 85 F.

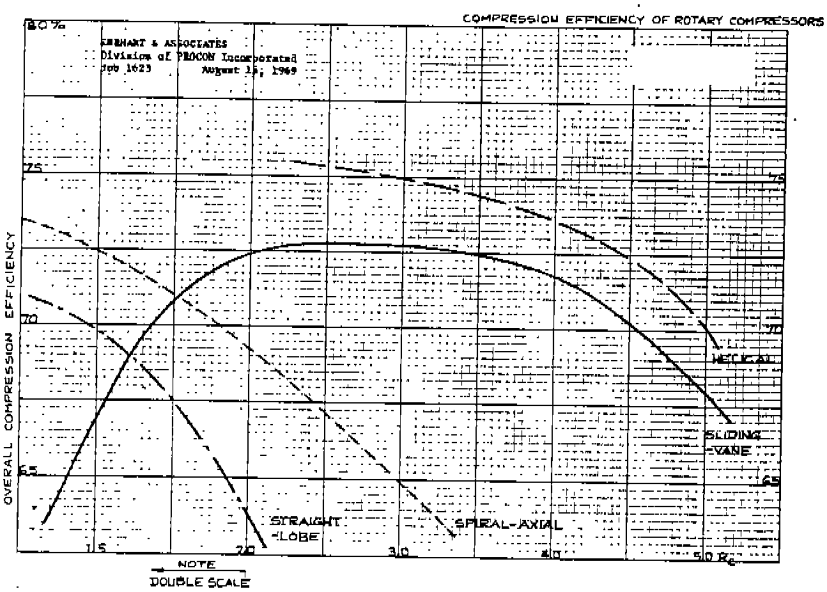

Fig.10 Compression efficiency of rotary compressor as compiled from catalog data

The liquid-liner rotary compressor has an average composite adiabatic and leakage efficiency of 45 percent at the normal rated tip speed of 66 fps. This efficiency is improved to 63 percent by reducing the tip speed to 30 fps and reduced to 40 percent at 80 fps. Judging from the data in Table 1, the charge and exhaust loss should not exceed 2 percent. The by-pass loss is estimated to be 3 percent. The mechanical losses should not exceed 5 percent.

NOMENCLATURE

AD = adiabatic function
A_e = rotor displacement form factor
bhp = brake horsepower
B = intrinsic R_c factor
cfm = cubic feet per minute
D_s = specific diameter $D\,L^{0.25}/Q^{0.5}$
d = diameter of rotor, in.
E_{vr} = volumetric efficiency for rotary units, percent
F = flow coefficient, dimensionless
fps = feet per second, velocity
G = gap of clearance, in.
icfm = inlet flow rate, cfm
k = ratio of specific heats, C_p/C_v
L = length of rotor, inches
m = molecular weight of gas
M* = Mach number, referred to sonic velocity, dimensionless
N_s = specific speed, $N\,Q^{0.5}/L^{0.75}$
N = speed of rotation, rpm
P = gauge pressure of gas system
P_1 = suction pressure, psia
P_2 = discharge pressure, psia
Q = volume flow, usually cfs
R = Rankine absolute temperature
R = ratio of compression
rpm = revolutions per minute
σ = $(k - 1)/k$
T = absolute temperature, degrees Rankine
U = tip-speed, fps
W = flow-rate, lb/min
W_L = labyrinth leakage, lb/min
v_s = specific volume, 10.73 Z T/m P
V = a volumetric constant
X = a displacement constant
X_t = thermal-leakage "warm-up"
y = decimal percent of Mach

SONIC-SLIP vs CLEARANCE EXPANSION

Conventional Evr/R curves can be drawn from empirical data taken from a vane compressor. One technician drew a composite Evr curve which has the characteristic of a 22% clearance cylinder. Another technician, using the same data, drew a composite Evr curve which represented an 8% clearance cylinder. The interstitial space of the first cell beyond the discharge "cut-water" is 3%. This is an incongruous situation to draft a representative clearance expansion equation.

The aperture slippage at a choked sonic velocity appears to be more representative behavior than the clearance expansion concept. The power requirement substantiates this premise. Instances where Evr was 50% and less, the power requirement substantiated the SLIP concept with the greater power load in lieu of the reduced power requirement which the clearance expansion behavior would have provided.

POSTSCRIPT

The conventions of BASIC computer programing are used in this text. The asterisk* indicates a multiplication, the slash / indicates a division and A ↑ X, indicates that A is raised to the X power.

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